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SIMULATED-ALTITUDE INVESTIGATIONS OF PERFORMANCE OF TUBULAR AIRCRAFT OIL COOLERS

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SUMMARY

The effects of altitude on the heat dissipation and on the oil pressure drop of an unjacketed and a conventional jacketed aircraft oil cooler were investigated at altitudes up to 40,000 feet. The investigations covered a range of oil flows from 50 to 150 pounds per minute, cooling air flows from 80 to 500 pounds per minute, and oil-inlet temperatures from 180° to 260° F. A method of correlating and of predicting cooling-air pressure drops at altitude has been indicated.

The results of the investigation showed that for a given cooling-air weight flow the sole effect of altitude on the heat dissipation and on the oil pressure drop was the effect of air temperature; the pressure of the air did not affect oil heat dissipation or oil pressure drop. Thus studies at sea-level pressure are satisfactory for investigating the oil heat dissipation and oil pressure drop at altitude provided that altitude air temperature and cooling-air weight flow are used. For constant oil flow and oil-inlet temperature, congealing of the oil occurred at altitudes above 10,000 feet and produced an initial decrease of 5 to 10 percent of the maximum heat dissipation attainable at each altitude; but when congealing was well established, the heat dissipation within the practical range of air flows remained essentially constant and independent of the air flow. The oil pressure drop similarly stabilized after the onset of congealing and increased only slightly with subsequent increases in the air flow.

The experimental cooling-air pressure drops were satisfactorily correlated and agreed closely with the predicted values. Temperature changes of the air obtained from sea-level investigations with low-temperature air could be employed in conjunction with the analytically derived expression for pressure drop to predict air pressure drops at altitude.

INTRODUCTION

Previous investigations of preformance of aircraft oil coolers have consisted of the determination of the heat dissipation and the pressure-drop characteristics at sea-level conditions and of the effect of air temperature on congealing characteristics at fixed air flow, oil-inlet temperature, and oil flow. The effect of altitude on the oil heat dissipation and the oil pressure drop was assumed to be the effect of air temperature alone but no experimental verification of altitude performance exists. The existing information on congealing is incomplete because the effect of each operating variable has not been previously investigated.

An investigation has been conducted at the NACA Cleveland laboratory with two tubular aircraft oil coolers over a range of simulated altitudes up to 40,000 feet to obtain experimental data on noncongealing and congealing performance at altitude. This investigation, which was conducted during the winter of 1944 and 1945, consisted of the determination of the effects produced on heat rejection, pressure drop, and congealing characteristics by changes in altitude, oil flow, and air flow.

APPARATUS

The dimensions of the two oil coolers investigated are given in the following table:

	Cooler A, unjacketed	Cooler B, jacketed
Diameter of core, inches	13	13
Number of tubes (equivalent open)	1184	1370
Tube size, inches		
Outside diameter	0.268	0.268
Diameter across flat sides	.323	.323
Length	9.5	9.5
Wall thickness	0.006	0.006
Number of oil passes	8	8
Weight, dry, pounds	45.8	43.9
Cooling surface, square feet	62.3	74.0

A schematic diagram of the unjacketed cooler A is presented in figure 1. During normal operation, oil enters at the top of the cooler, flows through 15 vertical tubes into a header at the bottom of the cooler, and from the header passes into the core.

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A schematic diagram of the jacketed cooler B is presented in figure 2. Oil enters at the top of the cooler, passes around one quarter of the jacket, then divides and flows in parallel paths through the jacket and through an internal wet baffle. During normal operation the oil passes from these warm-up passages to the bottom of the cooler and from there into the core. Within the core the oil flows axially along the outside of the tubes; seven baffles parallel to the tubes direct the flow through the cooler. The air flows through the tubes. A relief valve is located at the top of the cooler and flow restrictions are present in the outlet of the wet baffle and in the jacket. When the pressure difference between the inlet and the outlet of the core is greater than 30 pounds per square inch, the oil flowing through the warm-up passages continues around the remaining portion of the external jacket and passes through the relief port to the cooler outlet.

The oil coolers were arranged for the investigation as diagrammatically shown in figure 3. The inlet and outlet cooling-air ducts were pipes of the same inside diameter as the cooler core. The cooling air for this investigation was supplied by the refrigerated-air system of the laboratory and was removed by the altitude-exhaust system. The air was brought to desired temperatures by electric heaters in the line. The cooling-air flow and the pressure were manually regulated by throttling valves upstream and downstream of the cooler; thus, the quantity and the pressure could be independently varied. The air flow was measured with an adjustable orifice installed in accordance with A.S.M.E. specifications.

The oil was circulated by a motor-driven pump having a capacity of approximately 25 gallons per minute. The oil flowed through an electric heater, which could raise the temperature to desired values. The oil flow was set with a calibrated rotameter downstream of the cooler and was checked by a weigh tank.

All temperatures were measured with iron-constantan thermocouples connected through a selector switch to a self-balancing direct-reading potentiometer. Three thermocouples located at a station 8 inches upstream of the oil cooler were used to measure the temperature of the air entering the cooler. Two sets of 16 thermocouples were used to measure the temperature of the outlet air. One set of thermocouples arranged as shown in figure 3 was located 10 inches downstream of the oil cooler to measure the temperature pattern of the outlet air; the second set of similarly arranged thermocouples was placed immediately downstream of a mixing device to measure the average outlet temperature of the cooling air. The mixing device was a convergent-divergent pipe, circular at both ends and rectangular at the throat. The oil-inlet temperature was measured with three thermocouples spaced across the diameter of the oil line 3 inches from the cooler inlet. The oil-outlet temperature was measured with three thermocouples arranged on a horizontal diameter and three on a vertical diameter at a station 6 inches from the oil outlet. Oil leaving the cooler passed through a mixing cup placed between the cooler and the station at which the outlet temperature was measured.

Air static pressures upstream and downstream of the cooler were measured at stations located 4 and 6 inches, respectively, from the cooler faces. At each station, four wall pressure taps spaced 90° circumferentially were connected through a piezometer ring to a mercury manometer. An 11-tube total-head rake of streamline shape was located in the air duct at each static-tap station and each tube was connected to a water manometer. The pressure drop of the oil across the cooler was measured with a water-over-mercury differential manometer. The oil used in this investigation was AN-1120 conforming to specification AN-VV-0-446.

CONDITIONS OF RUNS

The greater part of the investigation was performed on unjacketed oil cooler A. For this cooler, thermal and pressure-drop data were obtained at the following conditions:

Altitude (ft)	Air flow (lb/min)	Oil flow (lb/min)	Oil-inlet temperature (°F)
0 - 40,000	80 - 500	100	225
0 - 20,000	50 - 420	50	225
30,000	180	50 - 150	225
30,000	180	100	180 - 260

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The smallest and greatest air flows at each altitude were the flows that resulted in air pressure drops of about 1 and 15 inches of water, respectively. The temperatures and pressures of the entering air were those of NACA standard atmosphere at the simulated altitudes, except at sea level at which the inlet-air temperature was 100° F. Measurements were also made of the sea-level isothermal pressure drop of the air at 80° F.

For jacketed oil cooler B, thermal data were obtained at simulated altitudes of 20,000, 25,000, and 30,000 feet for a range of air flows from 76 to 432 pounds per minute, an oil flow of 100 pounds per minute, and an oil-inlet temperature of 230° F.

All temperatures, pressures, and flows were established and maintained by manually operated controls. Repeat runs were made for each data point to check stability of the operating conditions. Especial care was exercised in runs wherein congealing of the oil was present.

METHODS

The heat dissipated by the oil H_o and the heat absorbed by the air H_a were computed from the respective relations

$$H_o = W_o \, c_{p,o} \, (T_{o,1} - T_{o,2})$$

$$H_a = W_a \, c_{p,a} \, (T_{a,2} - T_{a,1})$$

All symbols are defined in appendix A.

The specific heat of the oil varies linearly with temperature between 0.458 Btu per pound per °F at 100° F and 0.556 Btu per pound per °F at 226° F. In calculating the heat rejection, the value of specific heat corresponding to the average oil temperature was used. A value of 0.241 Btu per pound per °F was used for the specific heat of the cooling air.

The unit heat dissipation H_u, which is used as a measure of over-all heat-transfer coefficient in oil-cooler studies, was computed from the following relation:

$$H_u = \frac{H_o}{\left(\frac{T_{o,av} - T_{a,1}}{100}\right)}$$

where $T_{o,av}$ is the arithmetic average of oil-inlet and oil-outlet temperature, and H_u is the Btu dissipated per minute per 100° F temperature difference between average oil and inlet-air temperatures.

The following general equation for the pressure drop of the air is derived in appendix B as equation (6):

$$\sigma_1 \Delta p = C_1 \left(\frac{\rho_1}{\rho_2} + 1\right) W_a^{1.8} + \left[(C_3 - C_2) + \frac{\rho_1}{\rho_2}(C_2 + C_4)\right] W_a^{2.0}$$

The constants C_1, C_2, C_3, and C_4 depend on the geometry of the exchanger. The constants are defined in appendix A. For unjacketed oil cooler A, the equation resolved to (derived in appendix B as equation (6b))

$$\sigma_1 \Delta p = -0.131 \left(\frac{\rho_1}{\rho_2} + 1\right) W_a^{1.8} - \left(0.268 \frac{\rho_1}{\rho_2} - 0.230\right) W_a^{2.0} \qquad (6a)$$

For correlating pressure drops at all altitudes, a single exponent for W_a was determined for the purpose of expressing this equation in the alternate form $\sigma_1 \Delta p = W_a^{n} \Psi (\rho_2/\rho_1)$. In order to determine the value of n, values of ρ_2/ρ_1 between 1.0 and 0.75 were successively substituted. For each value of ρ_2/ρ_1, the pressure drops of two air flows were calculated by means of this equation and a straight line was drawn between the pressure drops plotted on logarithmic coordinates. The slope of each straight line was the value of n for the corresponding value of ρ_2/ρ_1. The slopes for values of ρ_2/ρ_1 between 1.0 and 0.75 were found to deviate from the slope at ρ_2/ρ_1 equal to 0.80 by ±1 percent of this slope. The air-flow exponent 1.85, which was the slope of the straight line for ρ_2/ρ_1 equal to 0.80, was used to correlate the experimental pressure drops. The equation for $\sigma_1 \Delta p$ implied by the use of this exponent was

$$\sigma_1 \Delta p = W_a^{1.85} \Psi(\rho_2/\rho_1)$$

An explicit expression for Ψ was unnecessary for correlating the experimental data and was not found.

The experimental air pressure-drop data were correlated by plotting the experimental value of $\sigma_1 \Delta p / W_a^{1.85}$ against the corresponding experimental value of ρ_2/ρ_1. From this plotted correlation of experimental data, the experimental pressure drops were replotted in conventional manner on logarithmic coordinates as follows: For each value of ρ_2/ρ_1 the corresponding experimental value of $\sigma_1 \Delta p/W_a^{1.85}$ was selected from the experimental data-correlation curve. The values of $\sigma_1 \Delta p$ for two air flows were then calculated by the relation

$$\sigma_1 \Delta p = (\sigma_1 \Delta p/W_a^{1.85})_{exp} \, (W_a^{1.85})$$

and a straight line was drawn through the two values of $\sigma_1 \Delta p$ plotted on logarithmic coordinates.

RESULTS AND DISCUSSION

The results of heat-dissipation studies and oil and air pressure-drop studies are presented in figures 4 to 14. All of these data are for unjacketed oil cooler A except those given in figure 7, which are for jacketed cooler B. The thermal performance results on the jacketed cooler are similar to those on the unjacketed cooler and figure 7 is presented to illustrate the comparative performance. In the following discussion, congealing of the oil will be understood to mean a chilling sufficient to decrease the heat dissipation below the values that would be obtained if the heat-dissipation trend with air flow persisted in the form established at low air flows.

Heat dissipation. - A comparison between the heat dissipated by the oil and the heat absorbed by the cooling air is shown in figure 4, which indicates satisfactory agreement between the dissipated and absorbed heats. The heat balance was equal to or better than 95 percent for 87 percent of the data.

The unit heat dissipation with 100 pounds per minute of oil at an oil-inlet temperature of 225° F for a range of air flows at temperatures and pressures corresponding to altitudes up to 40,000 feet is shown in figure 5. For each altitude the unit heat dissipation initially increases as the air flow increases. When the air flow becomes sufficiently great, however, to cause chilling of the oil film along a large part of the cooling surface, the unit heat dissipation at each altitude above 10,000 feet decreases by about 5 to 10 percent of the maximum value corresponding to the altitude and then remains essentially constant as the air flow increases. For each air flow the unit heat dissipation decreases as the altitude increases. The maximum unit heat dissipation and the subsequent decrease and leveling-off at each altitude are attained at progressively lower air flows as the altitude increases. At altitudes of 5000 and 10,000 feet, the unit heat dissipation levels off but does not decrease as the air flow increases to high values in the practical range. This leveling-off is probably due to a decrease in the oil heat-transfer coefficient resulting from the chilling of the oil film near the tube surface as the air flow increases. The decrease in oil heat-transfer coefficient appears just adequate to offset the increase in air heat-transfer coefficient arising from increases in the air flow. For each air flow the unit heat dissipation at 35,000 feet is the same as at 40,000 feet. This equality of the unit heat dissipations, considered together with the equality of the ambient-air temperatures at 35,000 and 40,000 feet, indicates that, for each air flow, the sole effect of altitude on heat dissipation is the temperature effect. Thus sea-level studies with low-temperature air are satisfactory for investigating oil heat dissipation and oil pressure drop at altitude.

The total heat dissipations from which the unit heat dissipations of figure 5 were calculated are shown in figure 6. For clarity, the curves for altitudes of 15,000 and 20,000 feet have been omitted from figure 6; they are similar to the total-heat-dissipation curves for 25,000 feet and higher altitudes. Figure 6 indicates that the behavior of the total heat dissipation at constant altitude is similar to that of the unit heat dissipation at constant altitude, as may be expected from the fact that H_o and H_u differ only by the factor $\frac{\left(T_{o,av} - T_{a,1}\right)}{100}$, the value of which does not change substantially at a particular altitude over the range of air flows encountered. As the altitude is increased, for constant air flow at noncongealing conditions, however, the total heat dissipation differs from the unit heat dissipation in the respect that whereas the unit heat dissipation decreases, the total heat dissipation increases. The drop in air temperature accompanying increases in altitude causes the oil to congeal at successively lower air flows, so that the total heat dissipation for each altitude above 10,000 feet reaches a maximum at progressively lower air flows, and these maximum attainable total heat dissipations decrease with altitude (fig. 6). After the oil has congealed, the total heat dissipation resumes its similarity to the unit heat dissipation in being a decreasing function of altitude.

The total heat dissipations of jacketed 13-inch cooler B operating with 100 pounds per minute of oil at an inlet temperature of 230° F and altitudes of 20,000, 25,000, and 30,000 feet are shown in figure 7. A comparison of figures 6 and 7 indicates that the thermal behaviors of the jacketed and unjacketed oil coolers under both noncongealing and congealing conditions are similar. The difference in magnitude of the heat dissipations of the two oil coolers is largely due to the difference of 11.7 square feet of cooling surface in the two coolers and is, in part, due to the 5° F difference in the oil-inlet temperatures that prevailed in the respective experiments.

The variation of the unit and total heat dissipations of the unjacketed cooler with air flow of 50 pounds per minute of oil at altitudes up to 20,000 feet is shown in figures 8 and 9, respectively. The characteristics of the curves in these figures are similar to those of the unit and total heat-dissipation curves obtained at a flow of 100 pounds per minute of oil. The magnitudes of both the unit and total heat dissipations, however, are smaller at 50 than at 100 pounds per minute of oil, and changes in the shapes of the curves occur at lower altitudes and air flows.

Slight decreases and increases with air flow in the unit and total heat dissipations under congealing conditions for flows of both 50 and 100 pounds per minute of oil are shown in figures 5, 6, 8, and 9. These slight changes may be due to instability of the congealed oil film. Within the practical range of air flows, well-defined congealing occurs at altitudes as low as 10,000 feet for 50 pounds per minute of oil. The flatness of the curves of total heat dissipation for high air flows at 5000 feet and at sea-level for 50 pounds of oil suggests that for air flows sufficiently large by comparison with oil flow, some chilling of the oil film is present at all altitudes and oil flows.

The effect of oil flow on the unit heat dissipation of the unjacketed cooler A, operating with an oil-inlet temperature of 225° F and an air flow of 180 pounds per minute at a simulated altitude of 30,000 feet, is shown in figure 10. Data from figures 5 and 8 have also been plotted in figure 10 and approximate curves have been faired to present an estimate of the effect of altitude on the variation of unit heat dissipation with oil flow at constant oil-inlet temperature. The curve for 30,000 feet indicates that at constant air flow and oil-inlet temperature the unit heat dissipation is proportional to the oil flow within the oil-flow range from 0 to 100 pounds per minute. At an oil flow of 100 pounds per minute, the heat dissipation starts to increase more rapidly than in direct proportion to the rate of oil flow and then begins to level off. The 30,000-foot heat-dissipation curve of figure 6 can be used to obtain information on the condition of the oil at the point at which the inflection exists in the 30,000-foot altitude curve of figure 10. The oil film is congealed (fig. 6) at an air flow of 180 pounds per minute and an oil rate of 100 pounds per minute at an altitude of 30,000 feet so that figures 6 and 10 together indicate that when the oil film at the tube surface has congealed, the total heat dissipation is directly proportional to the oil flow. The abrupt increase in the heat-dissipation curve for 30,000 feet as the oil flow increases beyond 100 pounds per minute is probably due to thawing of the oil from the congealed condition that prevailed at the lower flow rates. The subsequent leveling-off of the unit heat dissipation is due to the fact that the principal resistance to heat transfer in this operating region is in the air passage, so that the decrease in oil resistance due to increasing oil flow produces only a small change in the over-all resistance to heat transfer.

The effect of variation of the oil-inlet temperature on the unit heat dissipation of the unjacketed cooler for 100 pounds per minute of oil at an altitude of 30,000 feet is shown in figure 11. At an oil-inlet temperature of about 225° F the curve exhibits an inflection, and figure 6 indicates that at an altitude of 30,000 feet an oil flow of 100 pounds per minute, and an oil inlet temperature of 225° F the oil film is congealed at an air flow of 180 pounds per minute. The portion of the curve in figure 11 to the left of the inflection point thus shows the variation of unit heat dissipation with oil-inlet temperature under congealing conditions, and the portion of the curve to the right of the inflection point shows the variation of unit heat dissipation with oil-inlet temperature under noncongealing conditions.

<u>Oil pressure drop.</u> - The effect of cooling air flow on the nonisothermal pressure drop of the oil is shown in figure 12 for unjacketed oil cooler A. The data for this figure were obtained from the same investigations as those from which the thermal data for figures 5 and 6 were obtained, so that the conditions at which congealing became appreciable can be recognized and the characteristics of the oil pressure drop under normal and under congealing conditions can be distinguished. At each altitude for constant oil flow and constant oil-inlet temperature such as were maintained in these investigations, the oil pressure drop under noncongealing conditions increases as the cooling air flow increases (fig. 12). This figure also shows that at constant air flow under noncongealing conditions the oil pressure drop rises as the altitude

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is increased at all simulated altitudes lower than 35,000 feet. The oil pressure drop at each air flow is the same at 35,000 feet as at 40,000 feet. This similarity indicates that the sole effect of altitude on oil pressure drop is the temperature effect. The pressure rise with increasing altitudes below 35,000 feet is due to the increase in the mean viscosity of the oil as the oil is increasingly cooled. The curves of this figure indicate also that, as the cooling air flow approaches the value at which congealing of the oil becomes evident, the oil pressure drop increases more rapidly than under noncongealing conditions and then levels off as the air flow further rises. This increase in pressure drop is due not only to the increase in the mean viscosity but also to the thickening and lengthening of the layer of cold oil at the cooling surface, which causes a reduction in the passage area available for the flowing oil. The rise in the oil pressure drop is probably not as great, however, as these two factors would effect if they were present alone because, as the cold layer thickens, the resistance to heat dissipation also increases and causes the bulk of the oil to remain hot and of low viscosity. When congealing is well established, the oil pressure drop changes little with substantial increases in air flow, as shown in figure 12. This behavior is probably due to the attainment of equilibrium between two opposing tendencies, that of the viscosity to increase in the chilled and thickening film, and that of the viscosity of the core to remain low because of decreased heat dissipation.

Air-pressure-drop correlation. - The correlation of all the cooling-air pressure-drop data obtained at various altitudes, air and oil flows, and oil-inlet temperatures for unjacketed cooler A is shown in figure 13. This correlation indicates that a particular value of ρ_2/ρ_1 uniquely determines the value of $\sigma_1\Delta p/(W_a)^{1.85}$, independently of the altitude, air flow, oil flow, or oil temperature for which the value of ρ_2/ρ_1 was obtained. The curve also shows that a single air-flow exponent was satisfactory for correlating the data throughout the encountered range of ρ_2/ρ_1. The fact that a single exponent is satisfactory implies that the principal factor in determining the exponent is the geometry of the exchanger. The value of $\sigma_1\Delta p/(W_a)^{1.85}$ increases as ρ_2/ρ_1 decreases because progressively greater pressure drops are required to pump the air through the passage as the increase in air velocity from passage inlet to outlet becomes a greater fraction of the inlet velocity due to decreases in ρ_2/ρ_1. Decreases in ρ_2/ρ_1 result from increases in $\Delta p/p_1$, in $\Delta T/T_{a,1}$, or in both these ratios. The values of $\sigma_1\Delta p/(W_a)^{1.85}$ predicted from the analysis in appendix B (equation (6a)) deviate from a curve through the experimental results by

4 percent of the experimental value at ρ_2/ρ_1 equal to unity and by 8 percent of the experimental value at $\rho_2/\rho_1 = 0.75$. This satisfactory agreement between experimental and predicted values indicates that temperature changes of the air obtained from sea-level investigations with low-temperature air could be employed in conjunction with the analytically derived expression for pressure drop to predict air pressure drops at altitude. 854

A plot of the experimental values of $\sigma_1 \Delta p$ against W_a on logarithmic coordinates for various values of ρ_2/ρ_1 is shown in figure 14. This figure was drawn in accordance with the pressure-drop characteristics implied by the correlation of figure 13. The parallelism of the lines reflects the fact that a single air-flow exponent was satisfactory for every encountered value of ρ_2/ρ_1. The substantial increase in $\sigma_1 \Delta p$ as ρ_2/ρ_1 decreases at a specific air flow reflects the increase in the quantity $\sigma_1 \Delta p/(W_a)^{1.85}$ with decreasing ρ_2/ρ_1 and indicates clearly that a satisfactory prediction of $\sigma_1 \Delta p$ requires that the density ratio ρ_2/ρ_1 be considered.

SUMMARY OF RESULTS

The results of the simulated-altitude investigation of the performance of two 13-inch tubular aircraft oil coolers indicate that:

1. At constant air-flow rate, changes in the oil heat dissipation and in the oil pressure drop were caused by changes in the air temperature alone as altitude changed. The pressure of the cooling air did not affect either the heat dissipation nor the pressure drop of the oil. Sea-level studies with low-temperature air are thus satisfactory for investigating oil heat dissipation and oil pressure drop at altitude.

2. At specific operating conditions of oil flow, oil-inlet temperature, and air flow, the heat dissipated by the oil per 100° F difference between average oil- and air-inlet temperature, (unit heat dissipation), which is a measure of the coefficient of heat transfer from oil to air, decreased as altitude increased. The actual heat rejection of the oil (total heat dissipation) at specific operating conditions of oil flow, oil-inlet temperature,

and air flow, increased as altitude increased for the operating conditions at which congealing was not present, and decreased as altitude increased for the operating conditions at which congealing was present. In all other respects, the unit heat rejection and the total heat dissipation behaved similarly as altitude increased. For both quantities, the respective maximum values corresponding to each altitude were attained at progressively lower air flows, and the values of the maximums decreased as the altitude increased. At each altitude above 10,000 feet, the unit and total heat dissipations increased with increasing air flow until the air flow was sufficiently great to cause congealing of the oil. At the onset of congealing, the unit and total heat dissipations decreased by about 5 to 10 percent of the maximum values corresponding to the particular altitude and then remained essentially constant with substantial increases in air flow. This behavior was characteristic of both the unjacketed and the jacketed 13-inch coolers on which the experiments were performed.

3. When the oil film congealed at the cooling surface, the unit heat dissipation was directly proportional to the oil flow for a constant oil-inlet temperature and weight flow of cooling air.

4. At constant oil flow and oil-inlet temperature, the oil pressure drop increased with increases in air flow until the onset of congealing, after which the oil pressure drop changed only slightly with substantial increases in air flow.

5. The experimental cooling-air pressure drops were satisfactorily correlated and agreed closely with the predicted values. Temperature changes of the air obtained from sea-level investigations with low-temperature air could be employed in conjunction with the analytically derived expression for pressure drop to predict air pressure drops at altitude.

Flight Propulsion Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, November 21, 1947.

APPENDIX A

SYMBOLS

The following symbols are used in the calculations:

A_f flow area in air passage, sq ft

a constant, $(1/L)(1/\rho_L - 1/\rho_0)$

b constant, $1/\rho_0$

c_p specific heat at constant pressure, Btu/(lb)(°F)

D hydraulic diameter of passage through which air flows, ft

F friction, (ft)(force lb)/lb fluid

g acceleration of gravity, ft/sec^2

H_a total heat absorbed by air, Btu/min

H_o total heat dissipated by oil, Btu/min

H_u $H_u = \dfrac{H_o}{\left(\dfrac{T_{o,av} - T_{a,1}}{100}\right)}$, unit heat dissipation of oil, Btu/(min) (100° F difference between $T_{o,av}$ and $T_{a,1}$)

L length of smooth air passage, ft

l distance from inlet of smooth passage, ft

n exponent of air flow in pressure-drop-correlation equation

p static pressure of air, in. water or lb/sq ft

r_1 ratio of free-flow area at face of cooler with flared tube ends to frontal area of cooler

r_2 ratio of free-flow area within passage to frontal area of cooler

T total temperature, °F

u velocity of air, ft/sec

W_a air flow, lb/min or lb/sec

W_o oil flow, lb/min

Δp static-pressure drop, in. water

ΔT temperature change of air across oil cooler, ^{o}F

μ viscosity of air, lb/(ft)(sec)

ρ density of air, lb/cu ft

σ_1 ratio of ρ_1 to standard sea-level density

Ψ function of ρ_2/ρ_1

Subscripts:

a air

av average

exp experimental

L exit from smooth passage

o oil

s sea level

0 inlet to smooth passage

1 cooler inlet

2 cooler outlet

The values of constants appearing in equations (1(a)) to (6) are calculated as follows:

$$C_1 = \left(\frac{1}{0.0765}\right)\left(\frac{C_1'}{2}\right) = -\,0.00359\frac{\left(\mu^{0.2}\right)_{av} L}{{A_f}^{1.8}\,D^{1.2}}$$

$$C_2 = \left(\frac{1}{0.0765}\right)\left(C_2'\right) = -\frac{0.0779}{A_f^{\,2}}$$

$$C_3 = \left(\frac{1}{0.0765}\right)\left(C_3'\right) = -\frac{0.0389}{A_f^{\,2}}\left[1.09 - r_2^{\,2} + \left(\frac{r_2}{r_1}\right)^2\left(0.51 - 0.43\ r_1\right)\right]$$

$$C_4 = \left(\frac{1}{0.0765}\right)\left(C_4'\right) = +\frac{0.0389}{A_f^{\,2}}\left[1 - r_2^{\,2} - \left(\frac{r_2}{r_1} - r_2\right)^2\right]$$

$$C_1' = -\frac{0.000549\ \left(\mu^{0.2}\right)_{av}}{A_f^{\,1.8}\ \ D^{1.2}}$$

$$C_2' = -\frac{0.00596}{A_f^{\,2}}$$

$$C_3' = -\frac{0.00298}{A_f^{\,2}}\left[1.09 - r_2^{\,2} + \left(\frac{r_2}{r_1}\right)^2\left(0.51 - 0.43\ r_1\right)\right]$$

$$C_4' = \frac{0.00298}{A_f^{\,2}}\left[1 - r_2^{\,2} - \left(\frac{r_2}{r_1} - r_2\right)^2\right]$$

APPENDIX B

PRESSURE-DROP-CORRELATION PARAMETERS IN FLOW THROUGH TUBES

Inasmuch as the air pressure drop is a function of the weight flow, the inlet density, and the change in density of the air, the parameters that correlate pressure drops under diverse operating conditions will reflect all these variables. An investigation of the manner in which the variables determine the pressure drop follows.

The equation of energy balance of a fluid subjected to heat transfer and friction in steady flow in a differential length of smooth passage is presented in reference 1, page 117, equation 7(b). For an oil cooler, the term involving differences of elevation may be neglected and the equation may be written as

$$dp = -\rho dF - \frac{\rho u du}{g} \tag{1}$$

For dF, its equivalent may be substituted in the form presented in reference 1, page 119, equations (8) and (9a) for turbulent flow. Upon making this substitution, together with subsequent substitution of the quantity W_a/A_f for ρu on the basis of the continuity equation, and upon using the proper conversion factor to put the pressure drop in units of inches of water, equation (1) becomes

$$dp = C_1' \, W_a^{1.8} \frac{dl}{\rho} - C_2' \, \dot{W}_a^{2} \frac{d\rho}{\rho^2} \tag{1a}$$

Although the expression denoted by C_1' involves the variable viscosity of the air, C_1' may be taken constant because the viscosity appears only to the two-tenths power and $\mu^{0.2}$ changes only slightly within the practical range of air temperature. Integration of equation (1a) along the passage length yields the relation

$$\Delta p_{0-L} = C_1' \, W_a^{1.8} \int_0^L \frac{dl}{\rho} + C_2' \, W_a^{2.0}\left(\frac{1}{\rho_L} - \frac{1}{\rho_0}\right) \tag{2}$$

In order to evaluate $\int_0^L \frac{dl}{\rho}$, the axial distribution of density may be taken to be hyperbolic. The supposition of a hyperbolic density distribution is equivalent to the assumption that the axial distribution of velocity is linear. Figure 1 of reference 2 indicates that the assumption of linear velocity distribution is satisfactory for ratios of passage pressure drop to entrance pressure above 0.75. Then

$$\rho = \frac{1}{al + b}$$

where

$$a = \frac{1}{L}\left(\frac{1}{\rho_L} - \frac{1}{\rho_0}\right)$$

$$b = \frac{1}{\rho_0}$$

Then

$$C_1' \, W_a^{1.8} \int_0^L \frac{dl}{\rho} = C_1' \, W_a^{1.8} \int_0^L (al + b)\, dl = C_1' \, W_a^{1.8} \, \frac{L}{2}\left(\frac{\rho_0 + \rho_L}{\rho_0 \, \rho_L}\right)$$

Equation (2) may then be written as

$$\Delta p_{0-L} = C_1' \, \frac{L \, W_a^{1.8}}{2 \, \rho_0}\left(\frac{\rho_0}{\rho_L} + 1\right) + C_2' \, \frac{W_a^{2.0}}{\rho_0}\left(\frac{\rho_0}{\rho_L} - 1\right) \qquad (2a)$$

It may be assumed with negligible error that $\rho_0 = \rho_1$ and $\rho_L = \rho_2$. Equation (2a) then becomes

$$\Delta p_{0-L} = \frac{C_1'}{2} \, \frac{L \, W_a^{1.8}}{\rho_1}\left(\frac{\rho_1}{\rho_2} + 1\right) + C_2' \, \frac{W_a^{2.0}}{\rho_1}\left(\frac{\rho_1}{\rho_2} - 1\right) \qquad (3)$$

Upon entrance into the passage, the static pressure decreases because of (1) a friction loss resulting from the sudden decrease in flow area (vena-contracta loss), (2) a conversion of static to velocity head, and (3) an energy requirement for establishing a turbulent velocity distribiton.

By means of equation (13) (p. 122), figure 52 (both of reference 1), and the expression presented in reference 3 for the energy required to establish a velocity distribution, it can be shown that the complete static-pressure drop at the inlet, is given by

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$$\Delta p_1 = C_3' \frac{W_a^{2.0}}{\rho_1} \tag{4}$$

Upon exit from the cooler, there is a static-pressure regain due to deceleration of the air and a loss due to enlargement of the flow area. The loss resulting from sudden enlargement is given in reference 1, equation (12) (p. 121). The net exit pressure drop is obtained by combining the two components.

$$\Delta p_2 = C_4' \frac{W_a^{2.0}}{\rho_2} \tag{5}$$

The over-all static-pressure drop is the sum of the components. Upon multiplying through by ρ_1/ρ_s and combining the terms of equations (3) to (5), the result is

$$\sigma_1 \Delta p = C_1 \left(\frac{\rho_1}{\rho_2} + 1\right) W_a^{1.8} + \left[(C_3 - C_2) + \frac{\rho_1}{\rho_2}(C_2 + C_4)\right] W_a^{2.0} \tag{6}$$

With the exception of C_1, which involves the viscosity of the air, the constants depend only on the geometry of the exchanger.

Equation (6) is not an explicit solution for $\sigma_1 \Delta p$ because the ratio ρ_1/ρ_2 involves Δp. It is convenient, however, to keep the equation in this form to predict $\sigma_1 \Delta p$ and to select a parameter with which to correlate experimental pressure drops.

A knowledge of the proper values of ρ_1/ρ_2 to assign at anticipated operating conditions is necessary in order to predict $\sigma_1 \Delta p$. The following relation may be used to estimate ρ_1/ρ_2:

$$\rho_1/\rho_2 = \left(1 + \frac{\Delta T}{T_{a,1} + 460}\right) \Big/ \left(1 - \frac{\Delta p}{p_1}\right)$$

The quantity ΔT is estimated from the expected heat rejection and must be taken positive or negative depending on whether the fluid is heated or cooled. The quantity $\Delta p/p_1$ may be neglected in obtaining a first estimate of ρ_1/ρ_2. When Δp has been determined on the basis of the approximate value of ρ_1/ρ_2, this value of Δp may be substituted in the expression for ρ_1/ρ_2 to obtain a refinement of the quantity ρ_1/ρ_2 (reference 4).

For correlating experimental pressure drops, equation (6) shows that for a particular value of ρ_1/ρ_2 or ρ_2/ρ_1, $\sigma_1\Delta p$ depends only on the weight flow. The approximate equality of the exponents 1.8 and 2.0 makes possible for each value of ρ_1/ρ_2 the determination of an exponent n that permits writing the right member of equation (6) in the form $W_a^{\,n}\ \Psi(\rho_2/\rho_1)$ over a defined range of W_a. Although both n and Ψ depend on ρ_2/ρ_1, the exponent n varies only slightly with substantial changes in ρ_2/ρ_1, and for any one exchanger a single value of n can be obtained for which the right member of equation (6) can be satisfactorily represented by an expression of the form $W_a^{\,n}\ \Psi(\rho_2/\rho_1)$ over the entire range of ρ_2/ρ_1 encountered with the exchanger. If the pressure drops of two air flows are computed by equation (6) for an arbitrary value of ρ_2/ρ_1 in the range anticipated, the slope of the straight line joining the pressure drops on logarithmic coordinates is a satisfactory value of n. Equation (6) can then be written in the modified form

$$\sigma_1\Delta p/W_a^{\,n} = \Psi(\rho_2/\rho_1) \tag{7}$$

Equation (7) indicates that the quantity $\sigma_1\Delta p/W_a^{\,n}$ is uniquely determined for a particular value of ρ_2/ρ_1, regardless of the combination of air flow, altitude, and heat-transfer conditions from which the value of ρ_2/ρ_1 results. This equation shows that ρ_2/ρ_1 is a satisfactory parameter for correlating pressure drops under diverse operating conditions. For the purpose of correlating experimental data, it is not necessary to know explicity the form of $\Psi(\rho_2/\rho_1)$; it is necessary to know only the exponent n. If n is known, the experimental data permit calculation of $\sigma_1\Delta p/W_a^{\,n}$ and ρ_2/ρ_1, and the curve of $\sigma_1\Delta p/W_a^{\,n}$ against ρ_2/ρ_1 is the graphical equivalent of $\Psi(\rho_2/\rho_1)$.

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For the unjacketed 13-inch oil cooler investigated, the values of the constants C_1, C_2, C_3, and C_4 were calculated to be -0.131, -0.435, -0.205, and 0.167, respectively, and equation (6) reduced to

$$\sigma_1 \Delta p = -0.131\left(\frac{\rho_1}{\rho_2} + 1\right) W_a{}^{1.8} - \left(0.268\,\frac{\rho_1}{\rho_2} - 0.230\right) W_a{}^{2.0} \qquad (6a)$$

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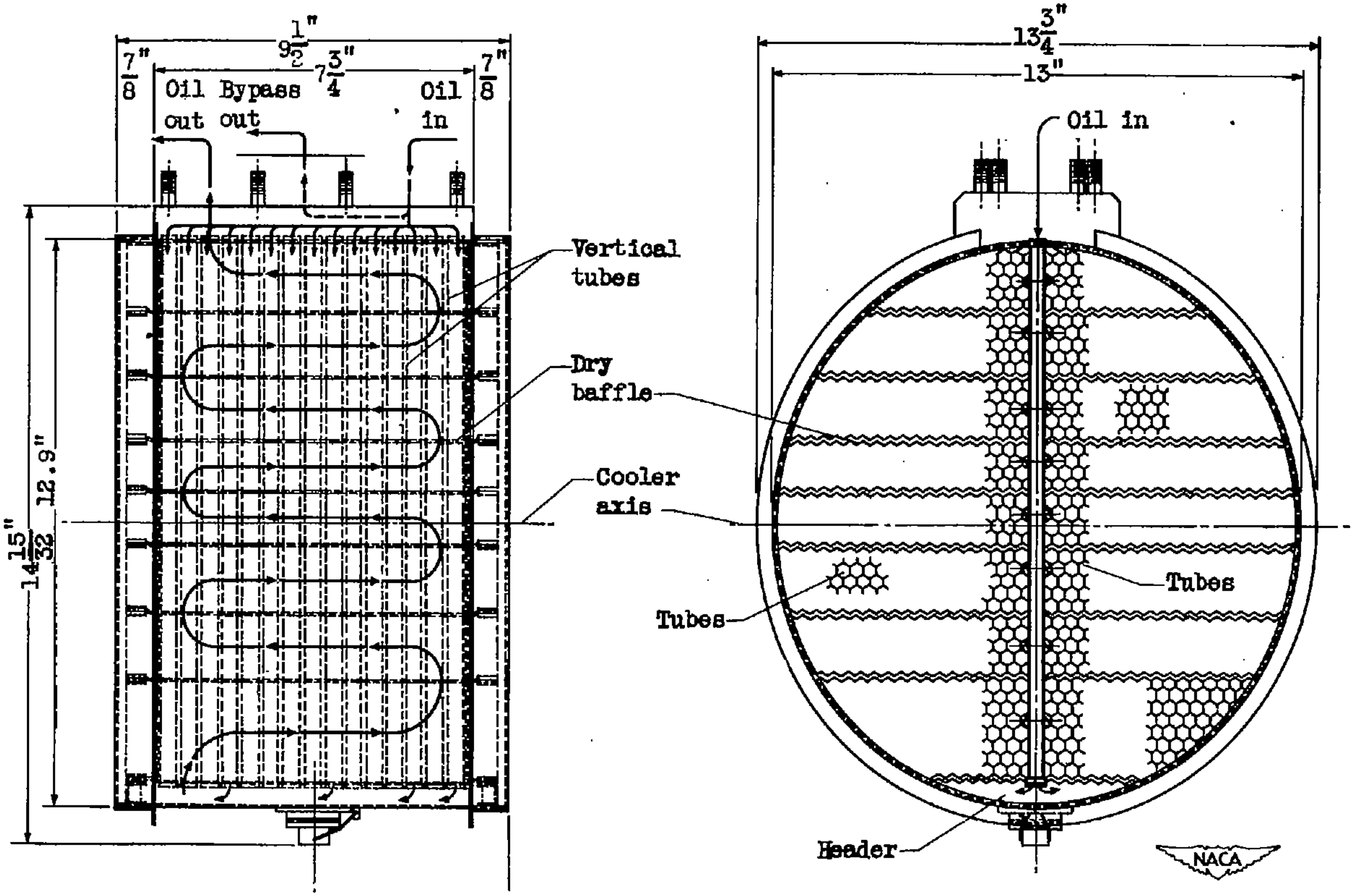

Figure 1. - Schematic diagram of unjacketed 13-inch oil cooler A.

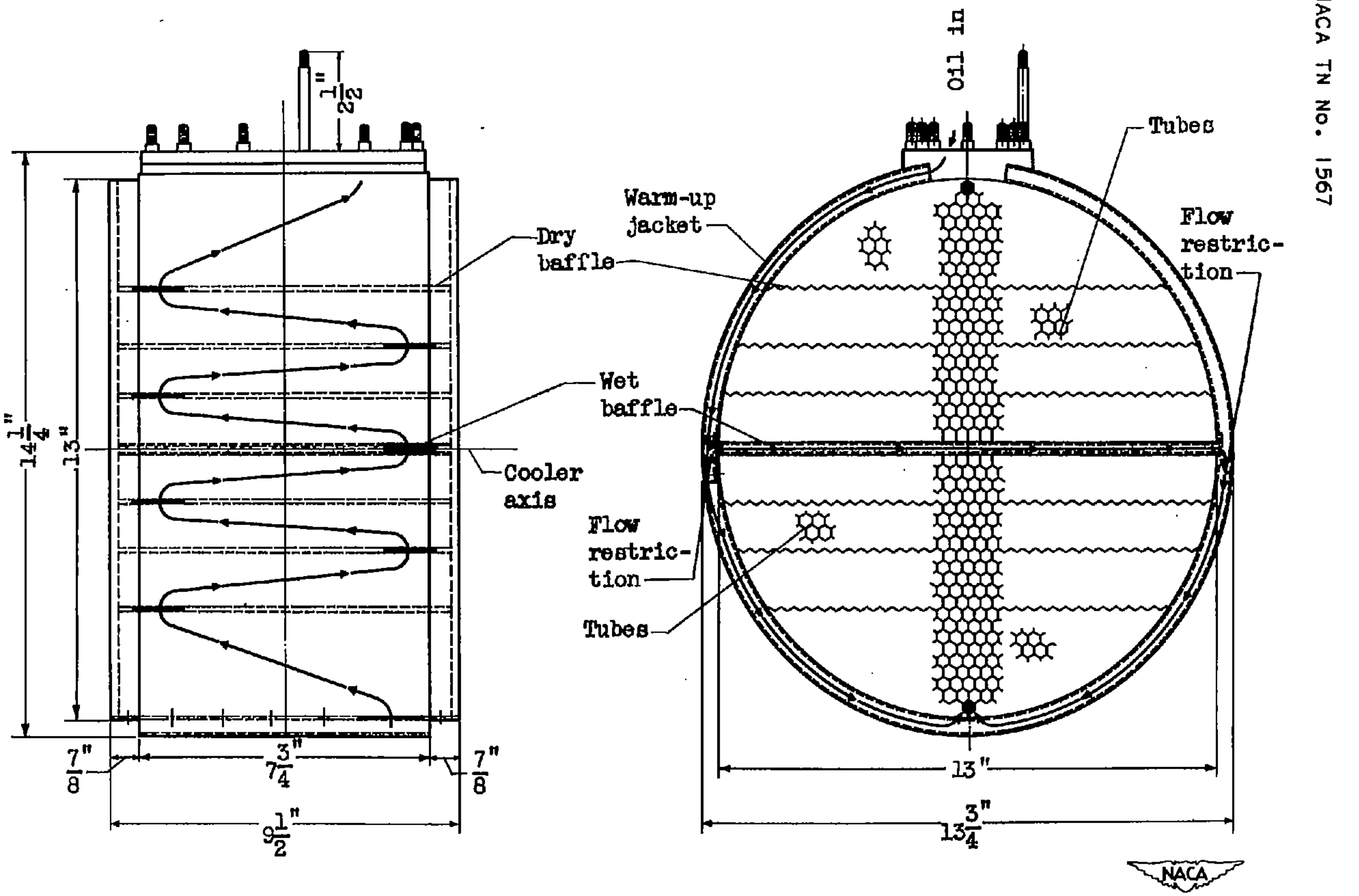

Figure 2. - Schematic diagram of jacketed 13-inch oil cooler B.

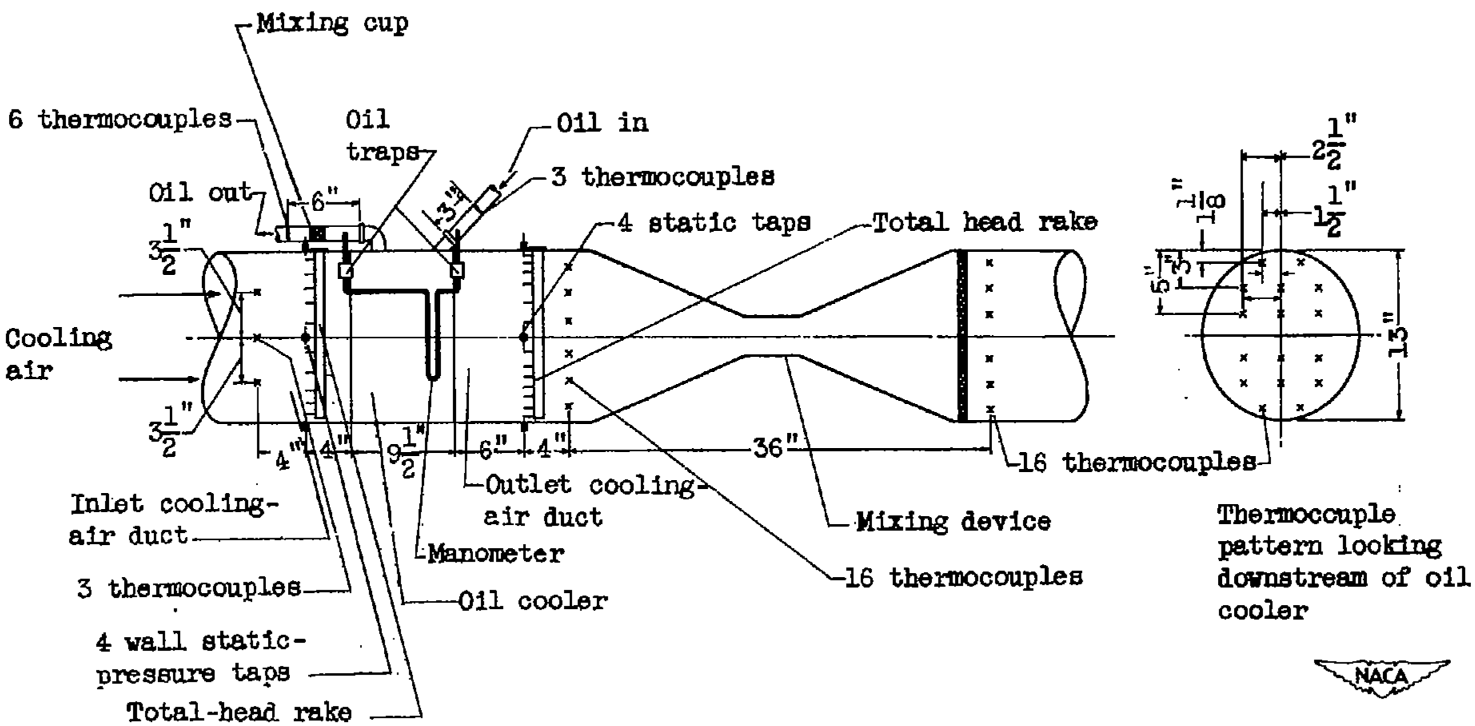

Figure 3. - Arrangement of oil-cooler setup.

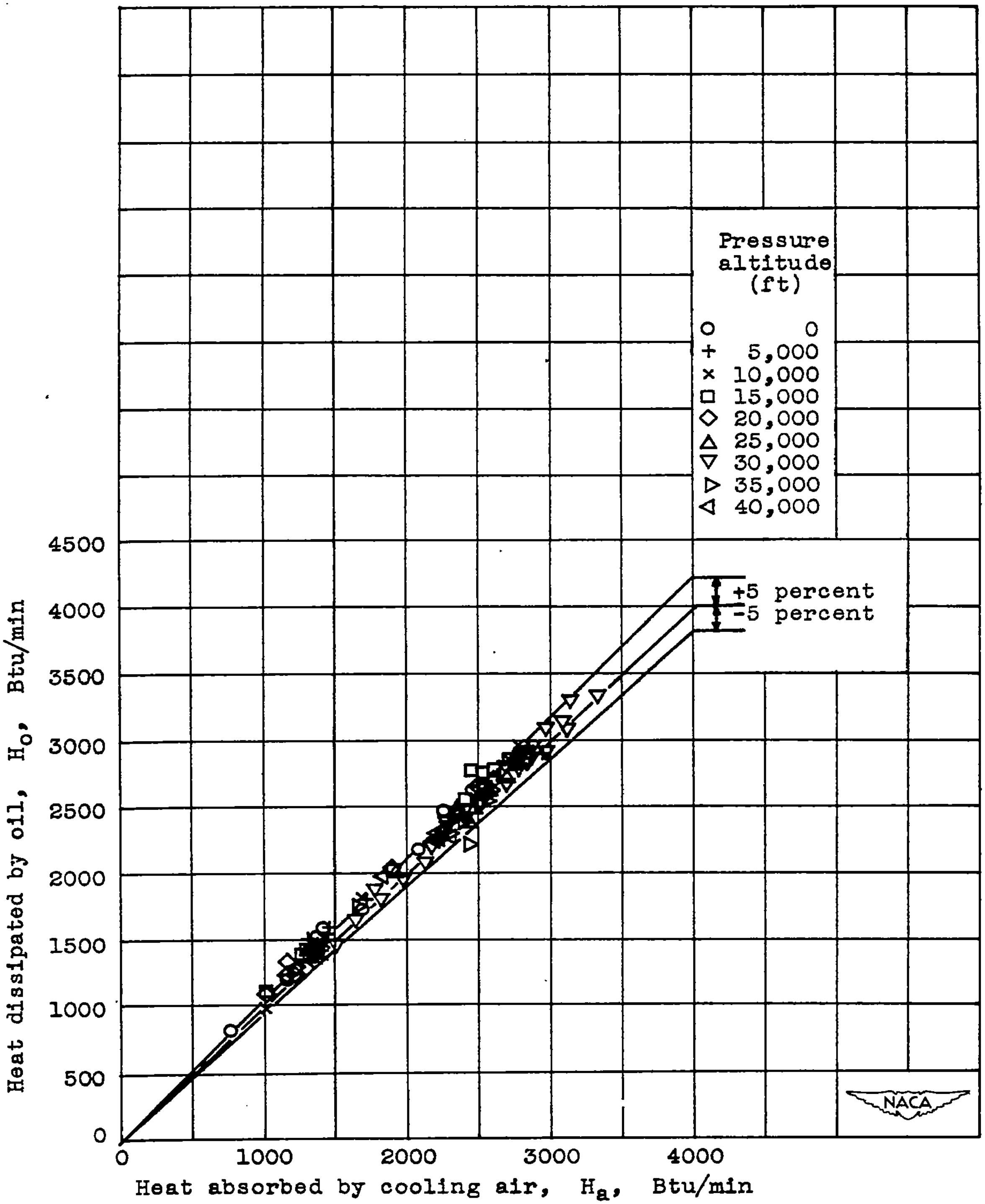

Figure 4. - Heat balance of oil and cooling air of unjacketed 13-inch oil cooler A.

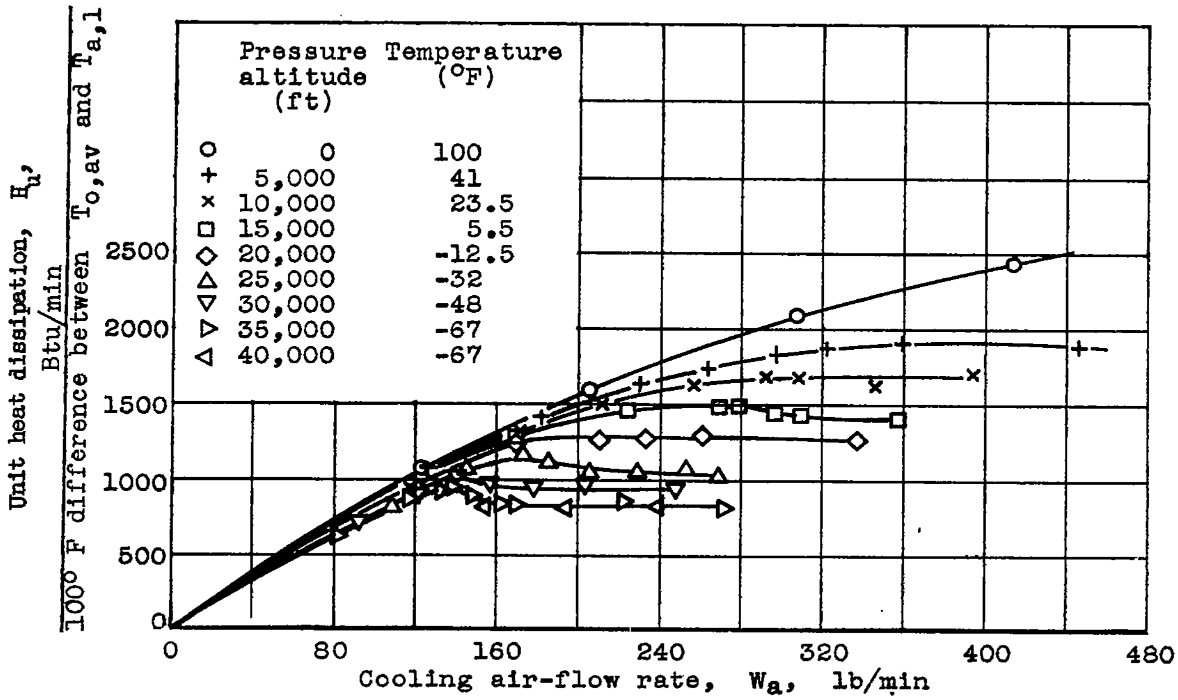

Figure 5. - Unit heat dissipation of unjacketed 13-inch oil cooler A. Oil flow, 100 pounds per minute; oil-inlet temperature, 225° F.

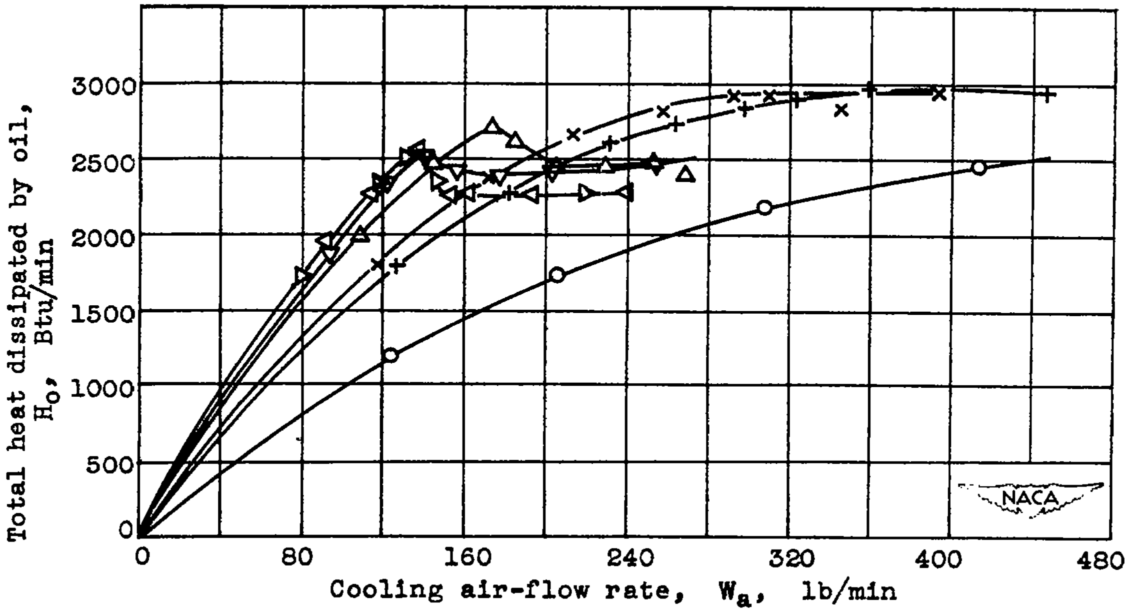

Figure 6. - Total heat dissipation of unjacketed 13-inch oil cooler A. Oil flow, 100 pounds per minute; oil-inlet temperature, 225° F.

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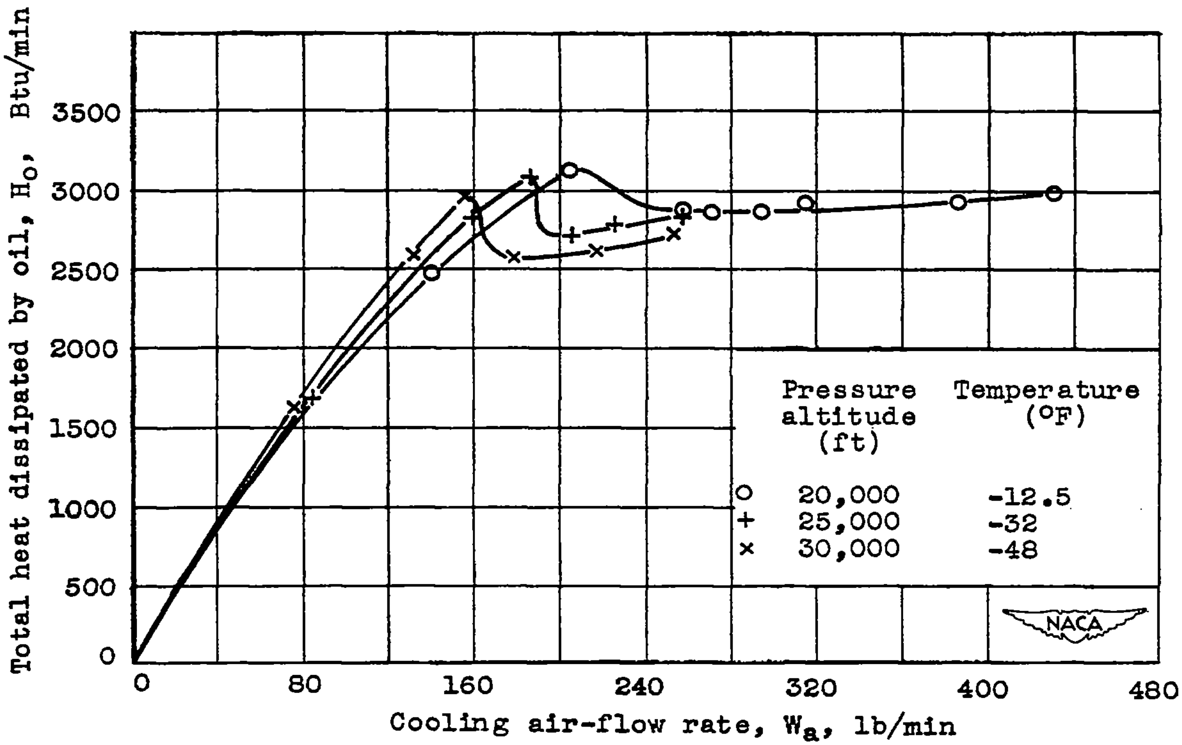

Figure 7. - Thermal behavior at altitude of jacketed oil cooler B. Oil flow, 100 pounds per minute; oil-inlet temperature, 230° F.

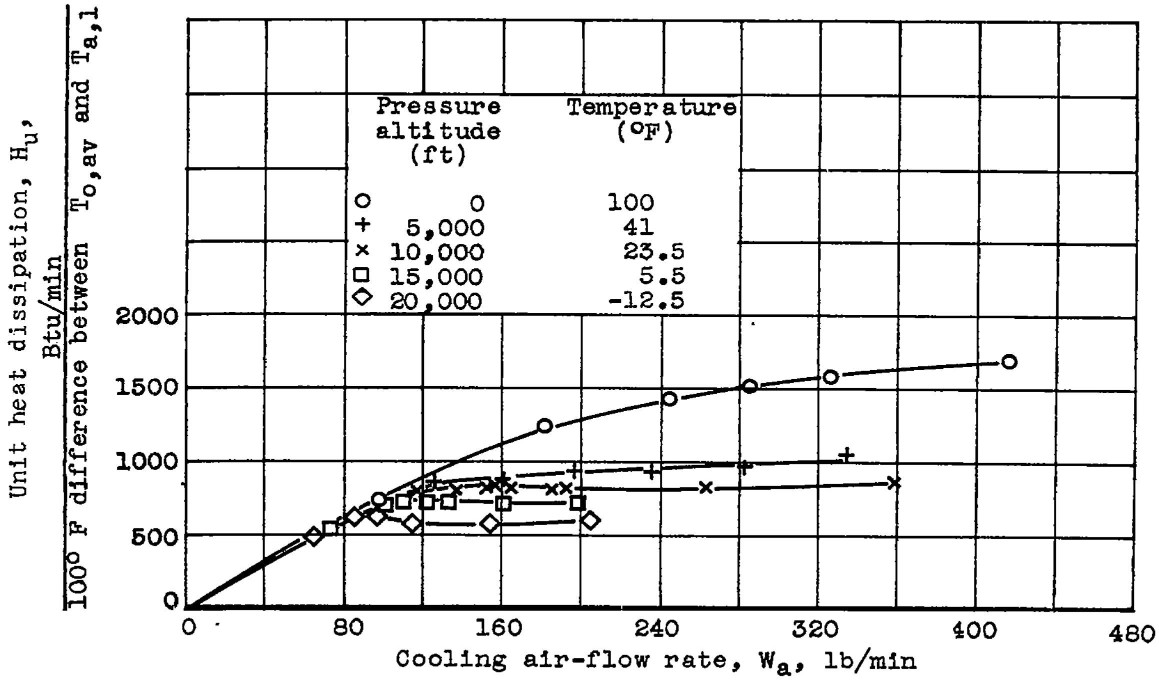

Figure 8. - Unit heat dissipation of unjacketed 13-inch oil cooler A. Oil flow, 50 pounds per minute; oil-inlet temperature, 225° F.

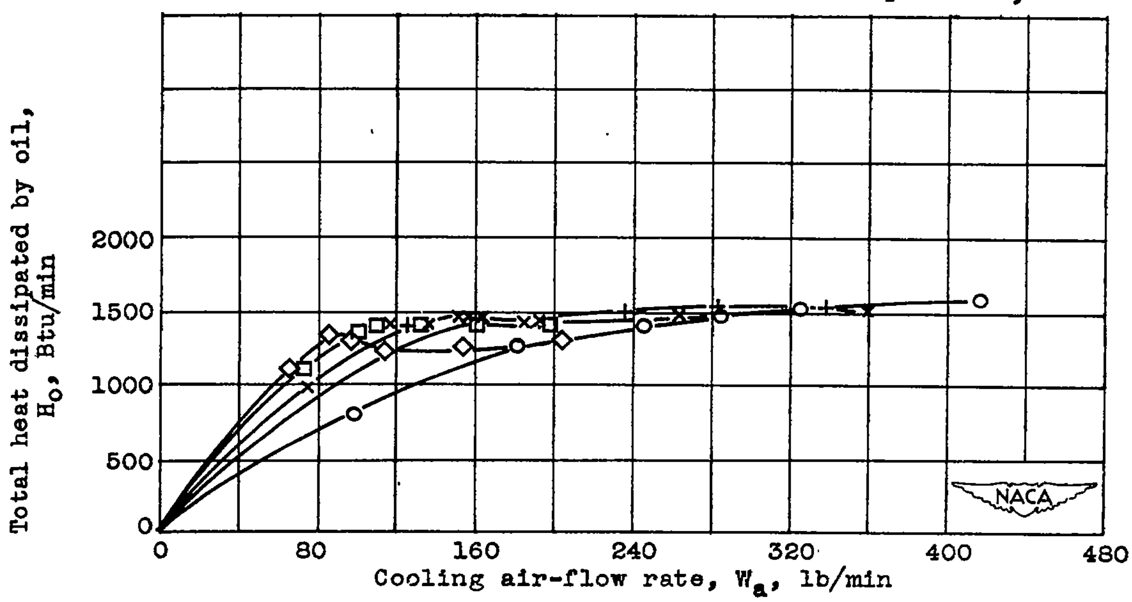

Figure 9. - Total heat dissipation of unjacketed 13-inch oil cooler A. Oil flow, 50 pounds per minute; oil-inlet temperature, 225° F.

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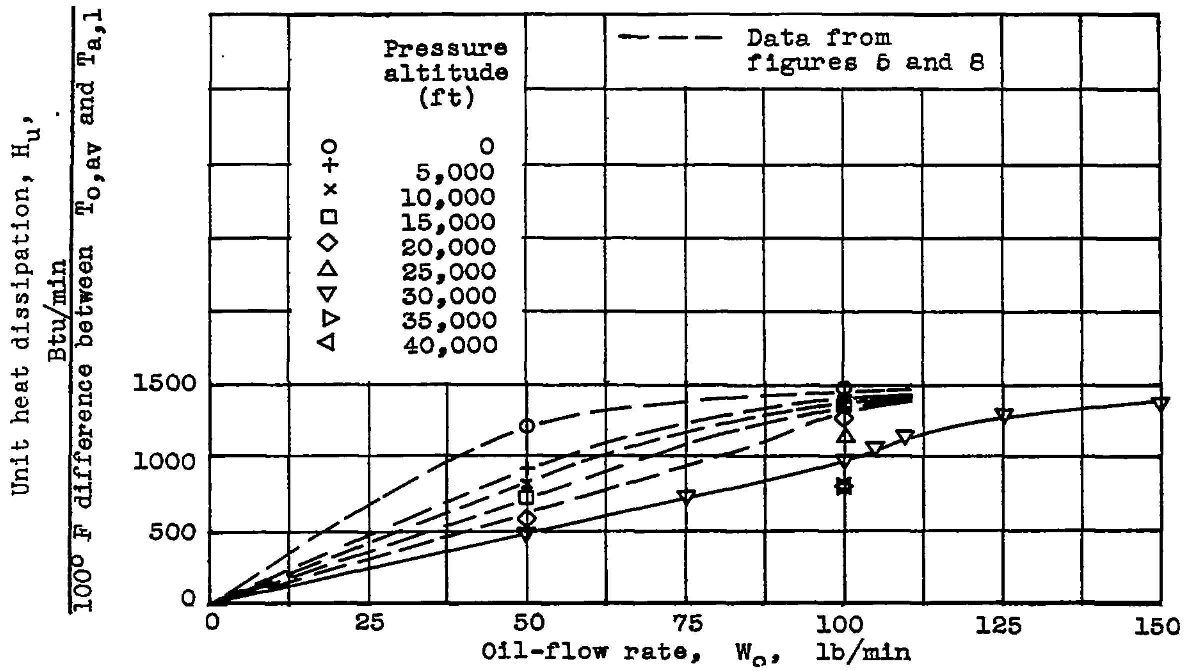

Figure 10. - Effect of oil-flow on unit heat dissipation of unjacketed 13-inch oil cooler A. Air flow, 180 pounds per minute, oil-inlet temperature, 225° F.

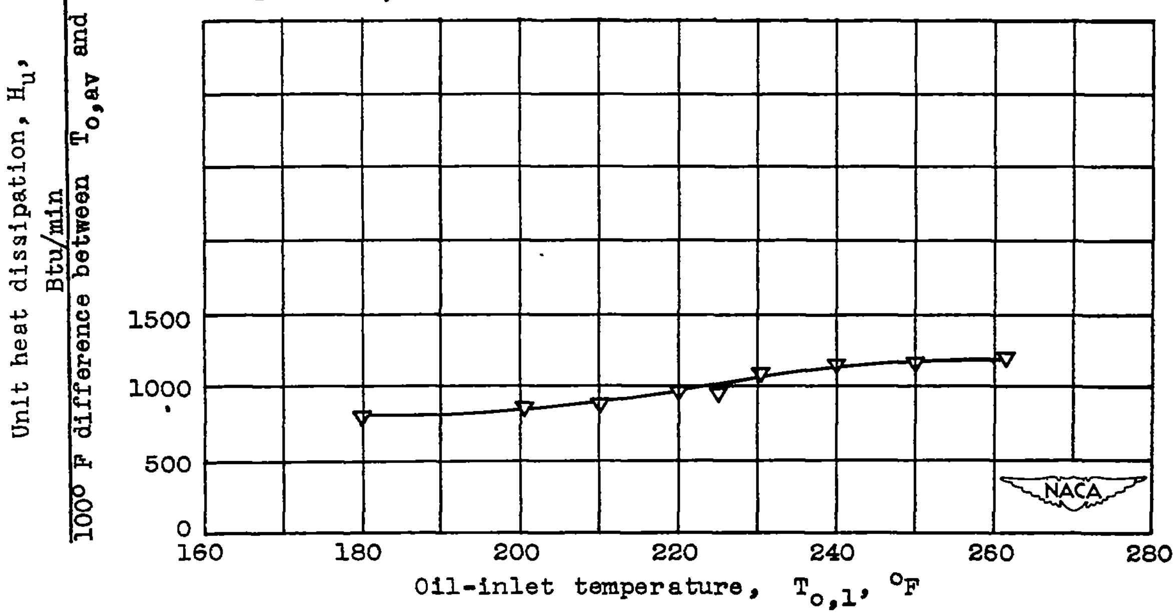

Figure 11. - Unit heat dissipation of unjacketed 13-inch oil cooler A as function of oil-inlet temperature. Oil flow, 100 pounds per minute; air flow, 180 pounds per minute; pressure altitude, 30,000 feet.

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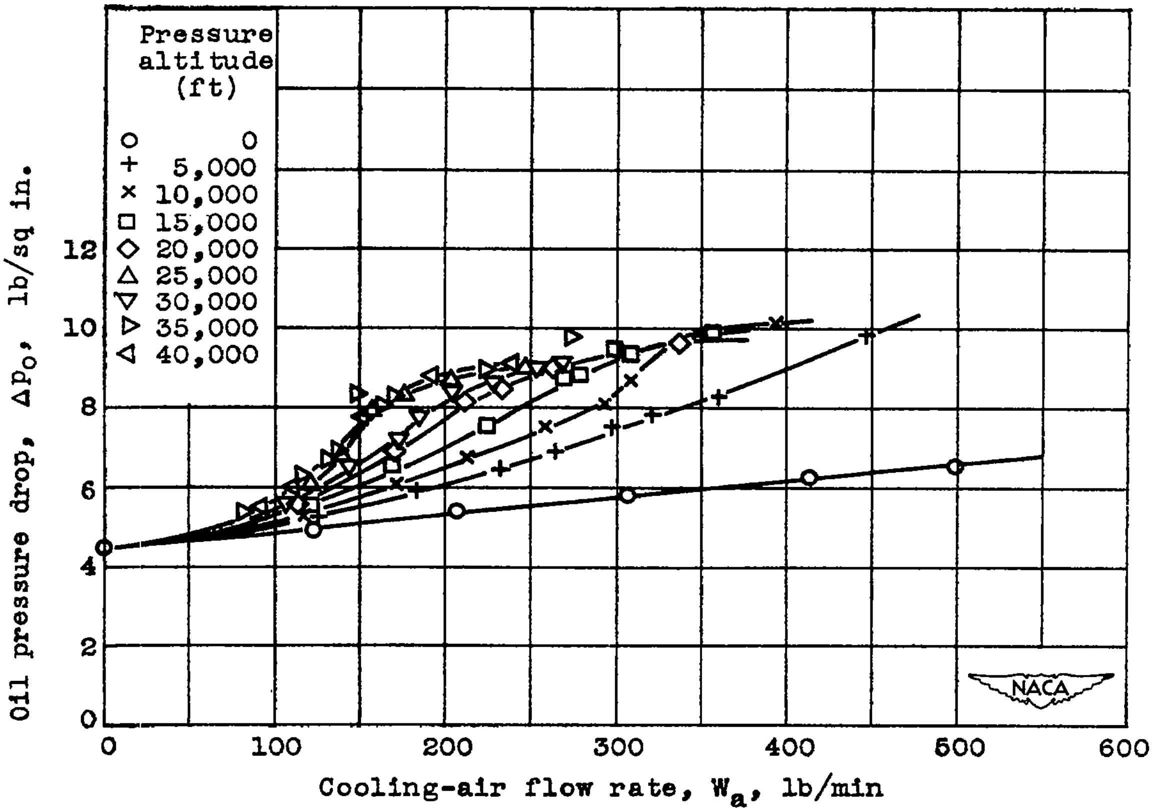

Figure 12. - Oil pressure drop of unjacketed 13-inch oil cooler A. Oil flow, 100 pounds per minute; oil-inlet temperature, 225° F.

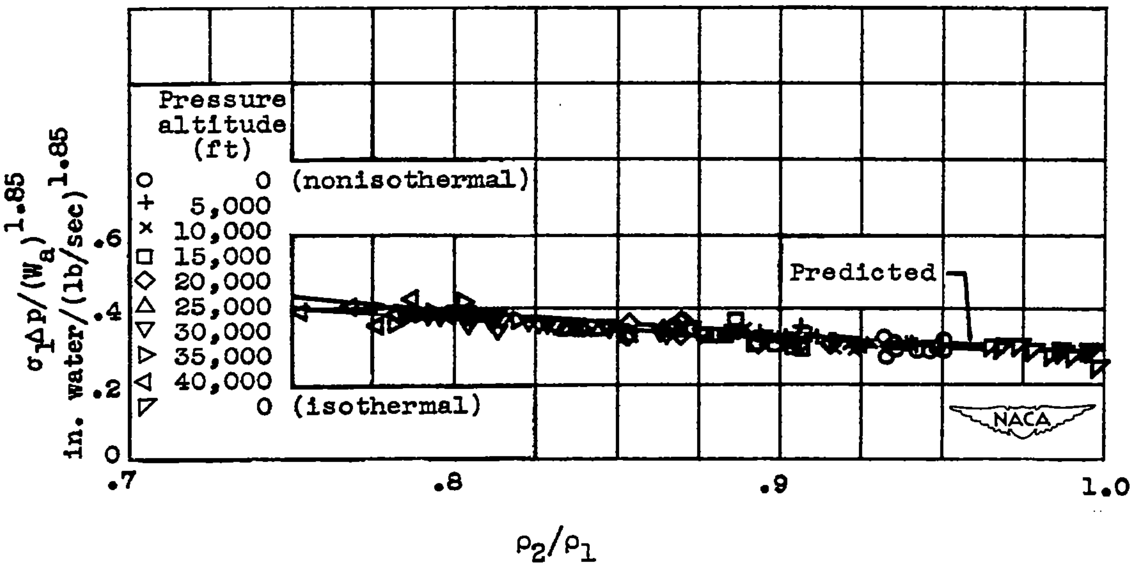

Figure 13. - Correlation of cooling-air pressure-drop data of unjacketed 13-inch oil cooler A.

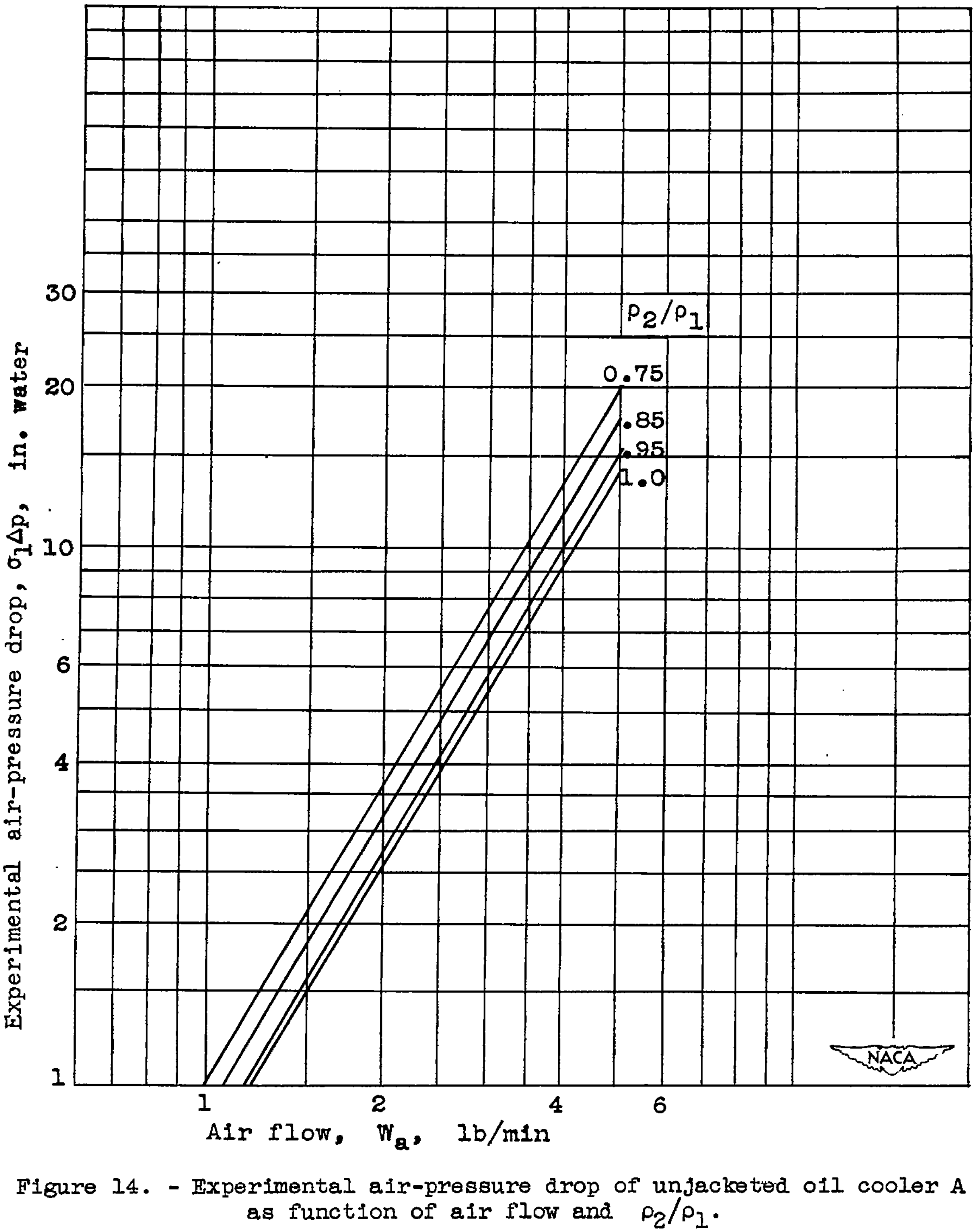

Figure 14. - Experimental air-pressure drop of unjacketed oil cooler A as function of air flow and ρ_2/ρ_1.